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SUMMARY

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An experimental investigation was undertaken to evaluate the [performance of two series of ball bearings (extra-light 108 and extremely light 1908) suitable for cryogenic turbopump applications.] These bearings were tested fully immersed in liquid oxygen at radial loads from 100 to 600 pounds and at DN values (bearing bore in mm times shaft speed in rpm) to 1.2 million. The maximum running time accumulated on any one bearing was 19.7 hours.

Bearing-torque and wear data were obtained on seven different retainer materials to compare their influence on overall bearing performance. Three different glass-reinforced-Teflon retainer materials that gave the best overall performance were (1) molybdenum disulfide plus glass fiber in Teflon, (2) glass fiber in Teflon, and (3) glass-cloth - Teflon laminate. Generally, the friction and wear results obtained corroborate earlier friction and wear studies in liquid nitrogen.

Low torque values and negligible wear characterized tests conducted on two 108-series bearings with molybdenum disulfide plus glass-fiber-in-Teflon retainers. Balls and races of these bearings were in excellent condition at the conclusion of each test series. Total test times on these two bearings were 8.5 and 6.3 hours.

Overall coefficients of friction obtained with bearings equipped with glass-fiber-in-Teflon retainers were of the same order of magnitude as those obtained by other investigators for oil-mist-lubricated ball bearings tested over the same radial-load range.

High rates of torque increase at shaft speeds exceeding 25 000 rpm were observed with the 1908-series bearings. High retainer forces characteristic of bearings running under combined loading probably distorted the lightly constructed retainers, which resulted in high-friction torques and retainer wear. These results are in contrast with published results on liquid-hydrogen-cooled 1908-series bearings operated at DN values to 1.6 million. Retainer forces generated by pure thrust loading on the bearing are less severe.

author

INTRODUCTION

Present generation liquid-chemical-propulsion systems utilize high-speed turbopumps to transfer cryogenic liquid propellants from storage tanks to the engine combustion chamber. Desirable design objectives for cryogenic pump systems could be achieved if it were practical to operate the shaft bearings immersed in the working fluid. Two advantages immediately apparent are (1) weight reduction by the elimination of a separate lubrication system and (2) simplification of design through the elimination of certain shaft seals required with conventional lubricants. To achieve maximum weight reduction and reduce complexity, it is necessary to use the working fluid to lubricate and cool the bearings. This report is concerned with research on ball bearings suitable for operation in liquid oxygen.

An ideal bearing lubricant basically performs two functions: (1) it minimizes wear and surface damage of the elements in sliding and/or rolling contact, and (2) it cools their surfaces. The first function is accomplished by the establishment of a surface-protecting boundary-lubricant film or by generation of an elastohydrodynamic film. The ability of a lubricant to generate an elastohydrodynamic film is proportional to its viscosity, and the absolute viscosity of liquid oxygen at -297° F (boiling point) is 0.027×10^{-6} reyn (0.186 centipoise) as compared with the absolute viscosity of an SAE 10 oil at 100° F of 4.9×10^{-6} reyn (33.8 centipoise). Liquid oxygen, therefore, can be expected to afford little protection at the ball-race contacts or at the retainer-ball and retainer-race contacts through the formation of any significant hydrodynamic films. Also, its effectiveness as a boundary lubricant, when compared with conventional organic lubricants, is very limited. The bearings must therefore incorporate some sort of self-lubricating mechanism. The second and most important function, when a cryogenic fluid is used as a lubricant, is to carry away the heat generated by frictional forces in the bearing and thus minimize the thermal gradients: these thermal gradients, when excessive, lead to a loss of internal clearance and seizure of the bearing.

The retainer materials selected for study in this investigation were self-lubricating materials that exhibit low wear and low coefficients of friction in cryogenic oxidants. Certain reinforced Teflon compositions have these desirable properties as determined by results of friction and wear studies (refs. 1 and unpublished NASA data) in cryogenic fluids. Also, results of tests on full-size bearings operated in liquid hydrogen (refs. 2 and 3) proved the superior performance of reinforced Teflon compositions over conventional metallic retainer materials.

The objectives of this report are to compare friction torque and wear in liquid oxygen for two series of 40-millimeter-bore ball bearings with different retainer materials at DN values (bearing bore in mm times the shaft speed in rpm) to 1.2 million. In the tests described herein, extra-light 108-series bearings were run at radial loads to 600 pounds and speeds to 30 000 rpm, and extremely light 1908-series bearings were run at radial loads to 300 pounds with a constant 25-pound thrust load and speeds to 30 000 rpm. Maximum anticipated radial-bearing loads in a typical oxidant turbopump would be in the range from 150 to 340 pounds at a shaft speed of 12 000 rpm. Bearing design geom-

Figure 1. – Lower half of test shaft and test-bearing assembly.

Labels (clockwise from upper left): Lower support bearing; Lower rotary face seal; Cover plate; Test-bearing housing; Thermocouple; Wave washer; Test bearing; Test shaft; Support-bearing housing; Oxygen vent; CD-7565

etries incorporated liberal radial clearances and open race curvatures for minimum heat generation (ref. 2).

APPARATUS

Drive System

The test apparatus used in this investigation had the test shaft mounted in the vertical position, the lower half of which is depicted schematically in figure 1. A variable-speed, direct-current drive motor was coupled to a speed increaser through a flexible gear coupling equipped with two shear pins to protect the drive train in the event of bearing seizure. The test bearing was located at the lower end of the test shaft that was mounted in a ball bearing and a roller bearing at the upper and lower ends, respectively. The test shaft was coupled at its upper end to the gearbox output shaft of the speed increaser by means of a special high-speed gear coupling. The speed of the gearbox output shaft was variable from approximately 400 to 50 000 rpm.
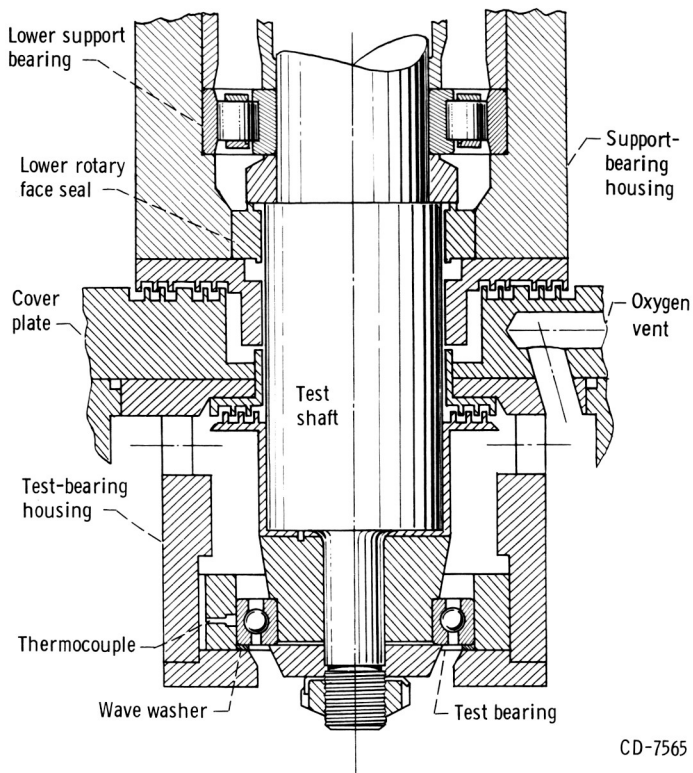
Load and Torque-Measuring Systems

Radial load was provided by a hydraulic cylinder and transmitted to the test bearing through a flexible steel strap (fig. 2). The drive system, test-bearing assembly, and their lubricating system were mounted on a steel bulkhead. This bulkhead pivoted independently of the load cylinder on upper and lower knife edge assemblies that had negligible friction compared with the test bearing. The entire weight of the assembly was supported by a 0.137-inch-diameter torsion rod. Test-bearing torque applied to its own housing rotated the entire assembly about the test bearing at the bottom and about the torsion rod at the top. A force transducer attached to the cover plate of the test-bearing housing measured test-bearing torque.

Cryogenic Supply System

The test chamber was a cylindrical weldment consisting of concentric chambers that enclosed the test-bearing assembly (fig. 3). The test chamber was attached to the cover plate and could be removed to provide access to the test bearing. The innermost chamber contained the liquid oxygen. The outermost
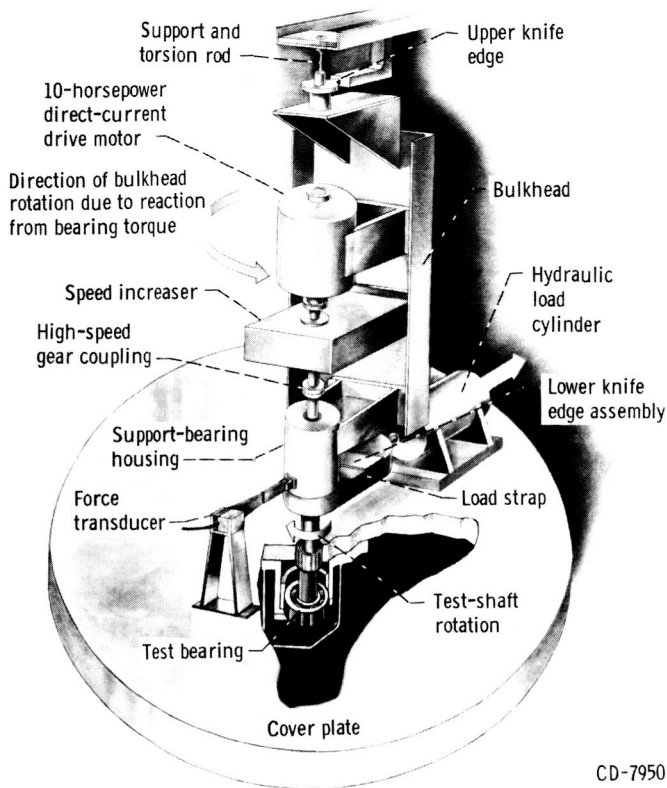
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Figure 2. - Schematic drawing of liquid-oxygen test apparatus.

chamber was a vacuum jacket for minimizing oxidant boiloff during a run. Liquid oxygen was stored in a 100-liter insulated Dewar vessel located adjacent to the rig. Transfer of the liquid oxygen from the Dewar to the oxidant chamber was accomplished by pressurizing with oxygen gas. A level of liquid sufficient to fully immerse the bearings was maintained by monitoring instrumentation and manual control. Boiloff gas was vented from the oxygen chamber through piping to the outside of the test cell.

Instrumentation

Test bearing and ambient bath temperatures were sensed with copper-constantan thermocouples and read out on self-balancing, recording, and indicating potentiometers. A capacitance probe in conjunction with a circular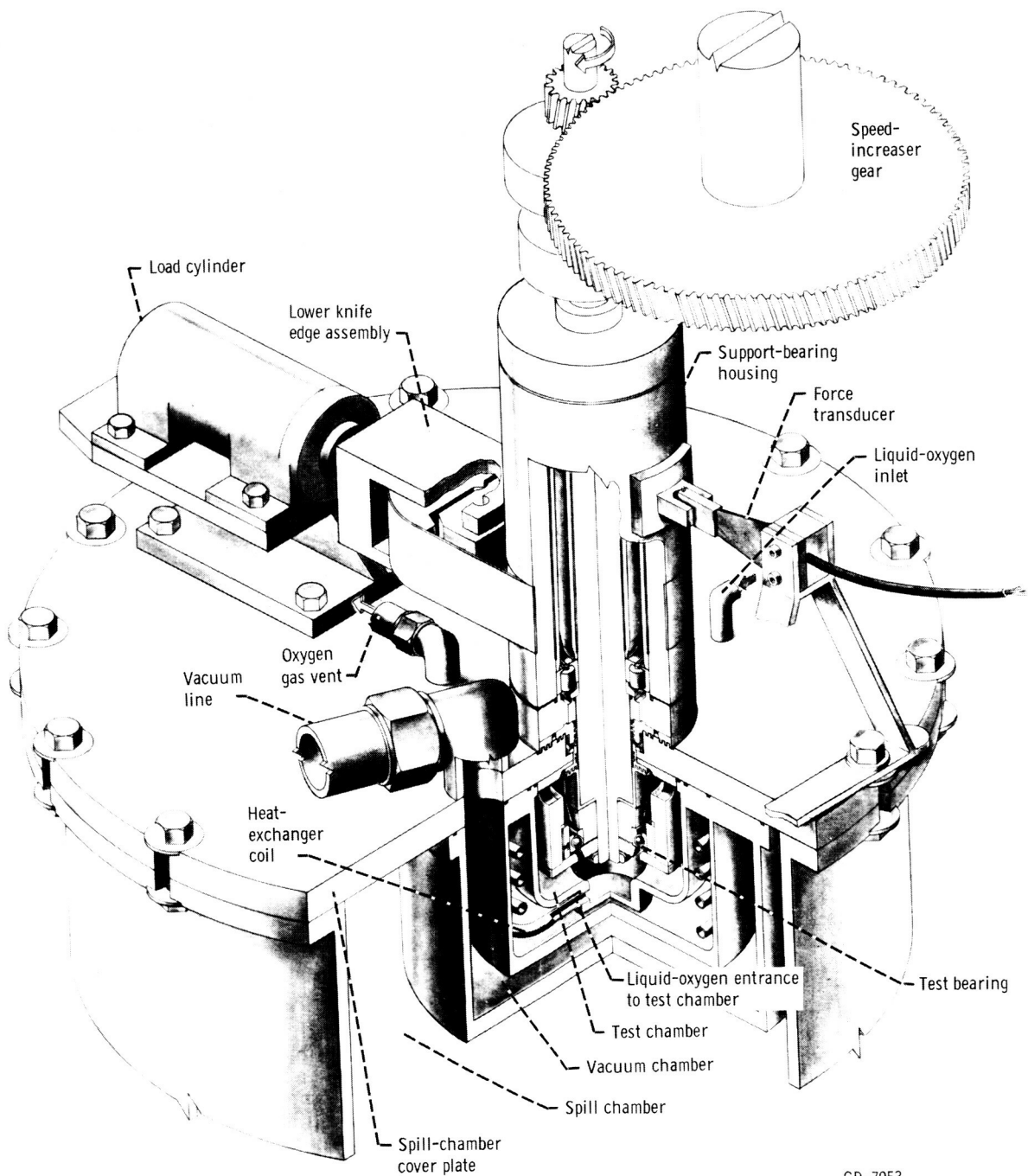-chart recorder provided a continuous record of the liquid-oxygen level during a test. Test-shaft speed was measured with a magnetic pickup head and displayed on a four-channel frequency counter.

Test Bearings

Light-series bearings were selected for testing in this program because radial loads imposed in oxidant turbopumps are not severe. To minimize corrosion, both races and balls were made of AISI 440-C stainless steel.

The 108-series, 40-millimeter-bore (1.575 in.) bearings had a ball complement of 10 and were of two types: (1) deep groove and (2) radial separable. The latter type was used in order to facilitate posttest inspection and weighing of components.

The 1908-series, 40-millimeter-bore bearings were all of the radial-separable type. Ball complements of 18, 14, and 12 were tested. Retainer bodies were reinforced with aluminum shrouds because of the low structural strength of the self-lubricating materials.

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Speed-increaser gear

Load cylinder

Lower knife edge assembly

Support-bearing housing

Force transducer

Liquid-oxygen inlet

Oxygen gas vent

Vacuum line

Heat-exchanger coil

Liquid-oxygen entrance to test chamber

Test bearing

Test chamber

Vacuum chamber

Spill chamber

Spill-chamber cover plate

CD-7953

Figure 3. - Liquid-oxygen bearing test rig.

PROCEDURE

Pretest Preparation

As-received test bearings were initially cleaned in a solvent (1,1,1 trichloroethane) and then washed in acetone prior to being placed in a vacuum desiccator from 14 to 16 hours. The bearings were then weighed on an analytical balance to the nearest 0.1 milligram. Each component of the separable bearings was weighed separately. After weighing, the radial play and the retainer locating clearance were measured, then the bearing was cleaned again in acetone and returned to the desiccator until installation in the test apparatus.

Test

After installation of the test bearing, the inner oxidant chamber was purged with helium gas for 10 minutes to ensure a minimum of moisture in the test chamber and on the test bearing. During the helium purge, the support-bearing and gear-lubricant pumps were turned on. At this time the torque measuring system was checked out.

The 100-liter Dewar was then pressurized with gaseous oxygen to transfer the liquid oxygen from the Dewar to the test chamber. The drive motor was started as soon as the liquid-level indicator and ambient-bath thermocouple indicated a liquid level sufficient to fully immerse the test bearing. At this time warm nitrogen gas was supplied to a series of labyrinth seals on the test-shaft housing to prevent the lubricating oil from freezing in the lower support-bearing housing. The test-shaft speed was then increased from approximately 750 to 2500 rpm, and the radial load was applied.

Two types of tests were conducted. One was a constant-load test with speed variable, and the second was a constant-speed load-variable test. For the variable-speed tests the shaft speed was increased in 2500-rpm increments up to 10 000 rpm and in 5000-rpm increments from 10 000 to 30 000 rpm with intervals between speed changes of approximately 5 minutes. The 108-series bearings were then operated at a speed of 30 000 rpm for 1 hour, and the 1908-series bearings for 1/2 hour. A test was terminated prematurely if the friction torque exceeded 3 pound-inches for a period of more than 5 minutes. For the variable-load tests the speed was held constant, and the load was varied in 100-pound increments up to 800 pounds. The time interval between load changes was sufficient to allow the friction torque to stabilize.

Posttest Inspection

After removal of the bearing from the rig, it was examined visually for any signs of distress. After flushing in acetone to remove any loose wear debris, the bearing was placed in the vacuum desiccator for 16 to 18 hours. The bearing was then weighed and the weight change recorded.

TABLE I. - TEST RESULTS IN LIQUID OXYGEN FOR EXTRA-LIGHT 108-SERIES BEARINGS

[Maximum DN, 1.2×10^6; bearing bore, 40 mm; shaft speed, 30 000 rpm; inner-race curvature, 0.53; outer-race curvature, 0.56; ball diameter, 3/8 in.; number of balls, 10.]

Bearing (a)	Retainer data			Total bearing radial play, in.	Radial-load range, lb	Time at maximum DN, hr	Total test time, hr	Comments
	Method of construc-tion	Material	Initial inner-race locating clearance, in.					
D-1	(b)	Glass-cloth - Teflon laminate	0.007	0.0009	50 to 500	2.6	19.7	All tests satisfactory; balls and races good; retainer fair
D-2	(b)	Glass-cloth - Teflon laminate	0.016	0.0008	200 to 500	4.9	9.3	High-load tests stopped at maximum DN because of high torque (fig. 10(a))
D-3	(b)	Glass fiber in Teflon	0.014	0.0010	100 to 500	2.3	17	All tests satisfactory; all components in good condition
D-4	(b)	Glass fiber in Teflon	0.014	0.0009	100 to 550	4.5	15.9	High-load tests stopped at maximum DN because of high torque; retainer not damaged (fig. 10(b))
R-1	(c)	Bronze powder in Teflon	0.024	0.0010	200 to 400	4	6	Test series stopped be-cause of excessive and nonuniform retainer wear
R-2	(c)	Bronze powder in Teflon	0.032	0.0008	200	0.6	1.9	Test series stopped be-cause of excessive and nonuniform retainer wear
R-3	(c)	Molybdenum disulfide plus glass fiber in Teflon	0.024	0.0008	200 to 600	3.6	8.5	All tests completed satis-factorily; all parts in excellent condition (fig. 12(a))
R-4	(c)	Molybdenum disulfide plus glass fiber in Teflon	0.024	0.0008	200 to 600	1.1	6.3	All tests completed satis-factorily; all parts in excellent condition
R-5	(c)	100-Percent graphite	0.011	0.0008	200 to 600	1.2	3.2	High-load tests stopped be-cause of excessive arma-ture current; balls and races severely damaged (fig. 12(b))

[a]D-lettered bearings are deep-groove type; R-lettered bearings are radial separable.

[b]Two-piece retainer body and two-piece aluminum shroud, riveted.

[c]One-piece retainer body, two-piece aluminum shroud, bolted.

RESULTS AND DISCUSSION

The results of tests on seven different retainer materials and three bearing types are given in tables I and II and in figures 4 to 14.

Effect of Speed and Load on Extra-Light 108-Series,

Deep-Groove Ball Bearings

Glass-cloth - Teflon laminate. - Satisfactory performance was obtained with this retainer material in bearings D-1 and D-2 (table I) at a maximum load of 500 pounds and 30 000 rpm (DN value of 1.2 million). At the maximum load and at a shaft speed of 30 000 rpm the Hertz maximum stress produced at the outer-race contact was 354 000 pounds per square inch.

TABLE II. – TEST RESULTS IN LIQUID OXYGEN FOR EXTREMELY LIGHT 1908-SERIES RADIAL-SEPARABLE BEARINGS

[Maximum DN, 1.2×10^6; bearing bore, 40 mm; shaft speed, 30 000 rpm; inner- and outer-race curvatures, 0.54; ball diameter, 1/4 in.; thrust load due to wave washer, 25 lb.]

Bearing (a)	Retainer data			Total bearing radial play, in.	Number of balls	Radial-load range, lb	Time at maximum DN, hr	Total test time, hr	Comments
	Method of construction	Material	Initial inner-race locating clearance, in.						
R-6	(b)	Glass fiber in Teflon	0.015	0.0018	18	100 to 300	3.5	11.7	Retainer poor; cracked through ball pockets (fig. 14(b))
R-7	(b)	Glass fiber in Teflon	0.011	0.0021	18	100 to 200 [c]100 to 800	1.2	4.8	Retainer good; fatigue spalls in outer race
R-8	(b)	Glass-cloth – Teflon laminate	0.018	0.0017	18	100 to 300	2.4	7.8	Outer race galled; balls pitted and spalled; retainer condition poor (fig. 14(a))
R-9	(d)	Bronze powder in Teflon	0.014	0.0020	12	------------	---	0.3	Unable to apply load; excessive torque; uneven wear
R-10	(d)	Molybdenum disulfide plus glass fiber in Teflon	0.024	0.0026	14	100 to 200	1.2	3.9	Aluminum shroud damage at ball pocket (fig. 14(c))
R-11	(d)	Molybdenum disulfide plus glass fiber in Teflon	0.018	0.0021	14	100	0.2	1.8	Aluminum shroud damage at ball pockets; retainer contacted outer race
R-12	(d)	100-Percent graphite	0.010	0.0022	12	100	0.5	1.4	Retainer cracked and broke out at ball pockets
R-13	(e)	Cotton-cloth-phenolic laminate	0.020	0.0016	14	100	---	0.7	Seizure at 27 500 rpm; retainer consumed by fire
R-14	(d)	Compacted molybdenum disulfide	0.026	0.0018	12	100	0.9	2.2	Excessive wear on retainer; balls contacting shroud; shroud contacting at outer race

[a]R-lettered bearings are radial separable.
[b]One-piece retainer body, two-piece aluminum shroud, riveted.
[c]Short-duration, load-variable test.
[d]One-piece body, one-piece aluminum shroud, one side spun over.
[e]Two-piece body, riveted, no shroud.

A gradual increase of torque with speed was observed for bearing D-2 in both the 400- and 500-pound load tests. After 23 minutes of operation at a 500-pound load and 30 000 rpm, there was a sudden and rapid rise in friction torque from 1.3 pound-inches to a value exceeding 3.0 pound-inches, the maximum of the potentiometer scale. The test was terminated after 5 minutes at this torque level in order to avoid a serious failure of the test bearing. No significant rise in bearing outer-race temperature was noted during these high-torque situations. It was necessary, however, to increase the flow of liquid oxygen to the test chamber to keep the bearing totally immersed. This indicated that liquid-oxygen boiloff due to frictional heat generated in the bearing was greater than normal.

Glass fiber in Teflon. – Test bearing D-3 was operated successfully at 30 000 rpm and a 500-pound load for a total time in two tests of 45 minutes.
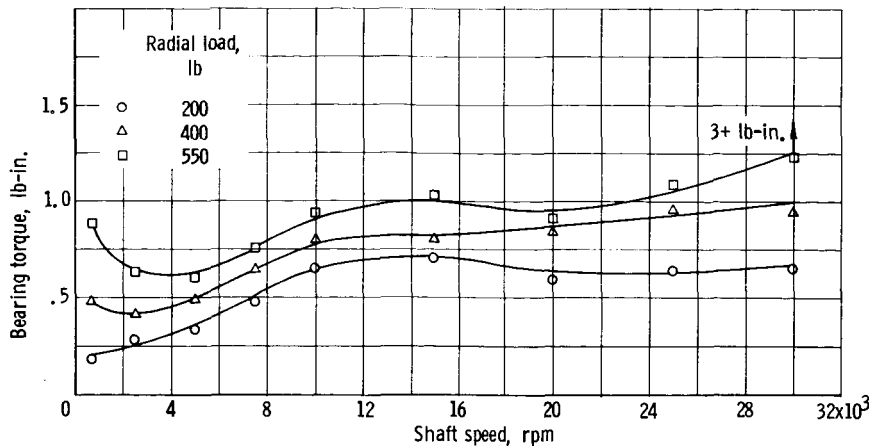
Figure 4. - Experimental bearing torque plotted against shaft speed for 108-series deep-groove ball bearings operating in liquid oxygen. Bearing, D-4; retainer material, glass fiber in Teflon.

Bearing D-4 also performed satisfactorily at a 550-pound load and 30 000 rpm for a total time of 32 minutes (test stopped after 1/2 hr because of a steadily increasing torque that eventually exceeded 3 lb-in.). A comparison of friction torque for bearing D-4 at varying speeds up to 30 000 rpm and three magnitudes of radial load is shown in figure 4. Generally, the torque-speed curves for this material are relatively flat. This agrees with experimental friction data for reinforced Teflon compositions in liquid nitrogen (ref. 1) and for full complement oil-lubricated ball bearings (ref. 4).

Effect of Speed and Load on Extra-light 108-Series

Radial-Separable Ball Bearings

Bronze powder in Teflon. - The torque values recorded for bearing R-1 at speeds to 30 000 rpm and at 400-pound radial loads were significantly higher than those measured on any of the bearings with glass-reinforced - Teflon retainers. Similar high-torque values were experienced with bearing R-2 at all speeds. Tests of this material were concluded prematurely because of excessive asymmetric wear at the retainer locating surface.

Molybdenum disulfide plus glass fiber in Teflon. - Test bearings R-3 and R-4 were operated successfully for 37 and 30 minutes, respectively, at a radial load of 600 pounds and shaft speeds to 30 000 rpm. Torque-speed data plotted in figure 5 show a magnitude comparable to that of the other Teflon compositions through 18 000 rpm (e.g., bearing D-4, fig. 4). The significant characteristic of these curves, however, is that the torque does not increase at higher speeds for the 400- and 600-pound load tests as does bearing D-4 at a 550-pound load. This would indicate that this retainer material established and maintained satisfactory boundary lubrication at maximum speed and load. Bearing and adapter parts were completely free of any wear debris. The ball set was uniformly dark in color, which indicated that the retainer material, which is black in color, was transferred to the ball surfaces.
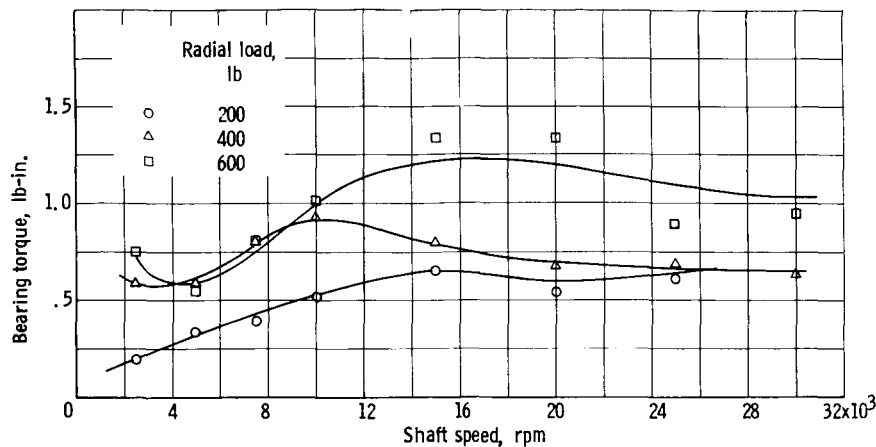
Figure 5. – Experimental bearing torque plotted against shaft speed for 108-series radial-separable ball bearings operating in liquid oxygen. Bearing, R-4; retainer material, molybdenum disulfide plus glass fiber in Teflon.

100-Percent graphite. - Satisfactory performance was obtained with this material in bearing R-5 at loads through 400 pounds. Friction torque over the speed range was stable and comparable in magnitude to that of the molybdenum disulfide plus glass-fiber-in-Teflon materials. After 35 minutes of operation at 30 000 rpm and a 600-pound radial load, however, the test was stopped because of an impending seizure, as indicated by excessive armature current, high-torque, and stalling of the motor. No further tests were attempted on this material because of the severe distress of balls, races, and retainers.

Effect of Speed and Load on Extremely Light 1908-Series,

Radial-Separable Ball Bearings

Glass fiber in Teflon. - Bearing R-6 (table II) was tested to a DN value of 1.2 million and at loads to 300 pounds. This combination of radial load and speed produce a Hertz stress maximum in the outer race of 296 000 pounds per square inch. A total time of 11.7 hours was logged on this bearing, approximately 3.5 hours of which were at the maximum speed of 30 000 rpm. Figure 6(a) compares torque values for bearing R-6 and a 108-series bearing (D-4) for loads of 300 and 400 pounds, respectively. The value of torque at speeds to 30 000 rpm and a 300-pound load for the 1908-series bearing exceeded that of the 108-series bearing running under a 400-pound load.

Glass-cloth - Teflon laminate. - Bearing R-8 performed satisfactorily at DN values to 1.2 million for the 100- and 200-pound loads, however, excessively high torque limited operation to 25 000 rpm (DN of 1.0 million) at a 300-pound load. As was the case with the glass-fiber-in-Teflon retainers, high rates of torque increase were observed at speeds exceeding 20 000 rpm. Figure 6(b) compares torque values for this bearing and a 108-series bearing, D-2, for loads of 300 and 400 pounds, respectively. Torque values for the 1908-series bearing are much higher at 30 000 rpm and 300 pounds than those of the 108-series bearing running at the same speed and at a 400-pound load. These results are in contrast to results reported with liquid-hydrogen-cooled bearings operated at

(a) Retainer material, glass fiber in Teflon.

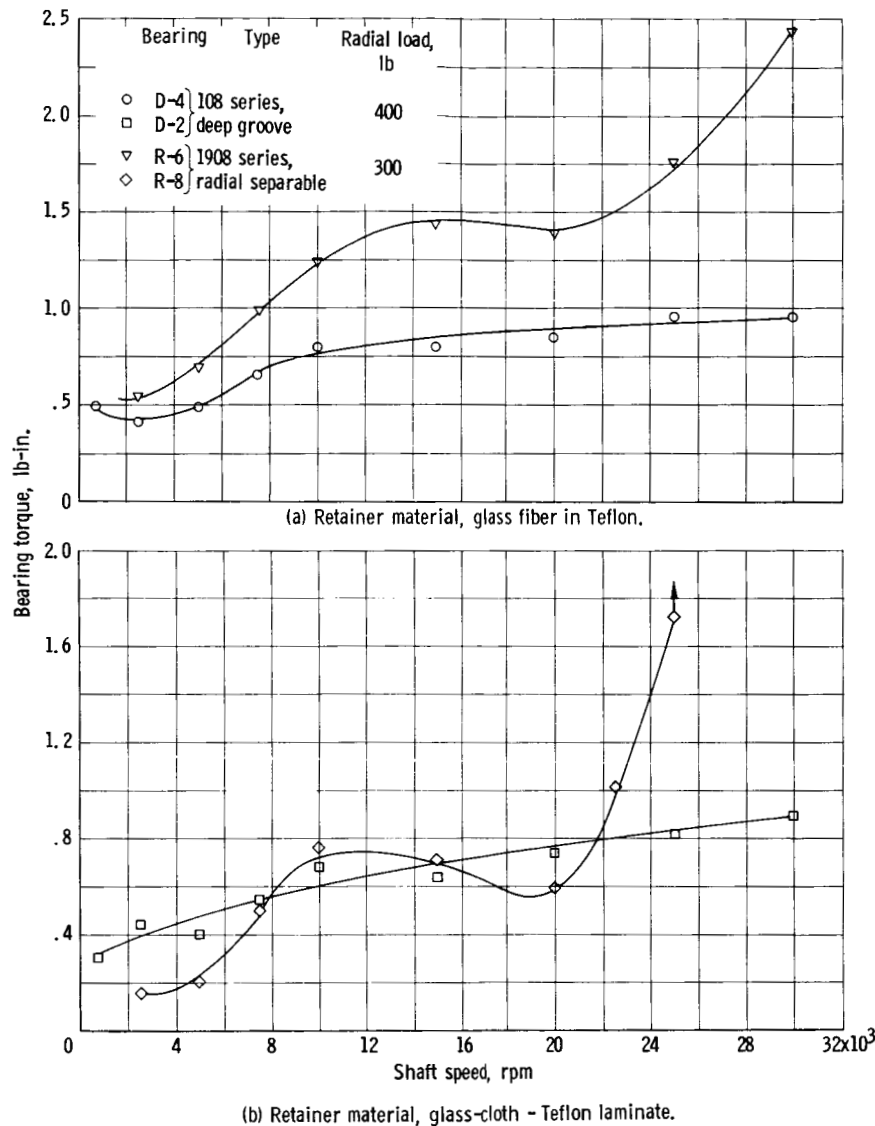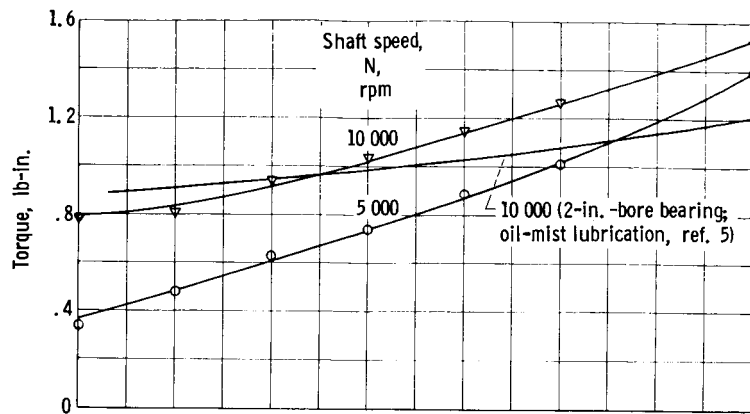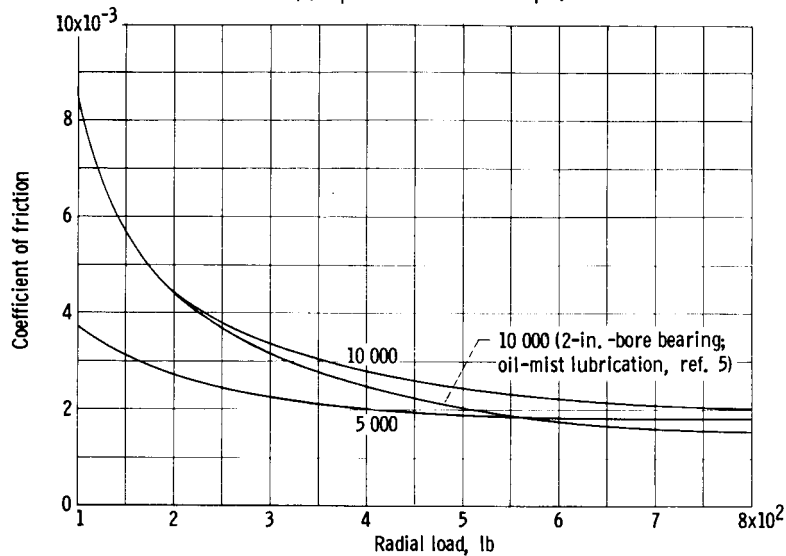(b) Retainer material, glass-cloth - Teflon laminate.

Figure 6. - Comparison of experimental bearing torque as plotted against shaft speed for 108-
and 1908-series bearings operating in liquid oxygen.

DN values to 1.6 million. The bearings tested in hydrogen, however, were operated under pure thrust loading, where retainer forces are generally not as severe.

Plotted in figure 7(a) are experimental values of friction torque against radial load for bearing R-7 with a glass-fiber-in-Teflon retainer. These values, as well as values of coefficient of friction (fig. 7(b)) are compared with data from reference 4 for an oil-mist-lubricated 2-inch-bore bearing operating at 10 000 rpm. A minimum coefficient of friction of 0.0020 was calculated for the liquid-oxygen-lubricated bearing at an 800-pound load and 10 000 rpm compared with a minimum value of 0.0015 for the oil-lubricated bearing at the same conditions of speed and load. These results indicate that ball bearings can operate in liquid oxygen with friction losses comparable to those of oil-lubricated bearings.

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(a) Experimental friction torque.

(b) Calculated friction coefficient.

Figure 7. – Comparisons of experimental friction torque and calculated friction
coefficients plotted against radial load for liquid-oxygen-cooled and oil-mist-
lubricated ball bearings. Bearing series, 1908; radial separable; bearing R-7;
retainer material, glass fiber in Teflon; race and ball material, AISI 440-C
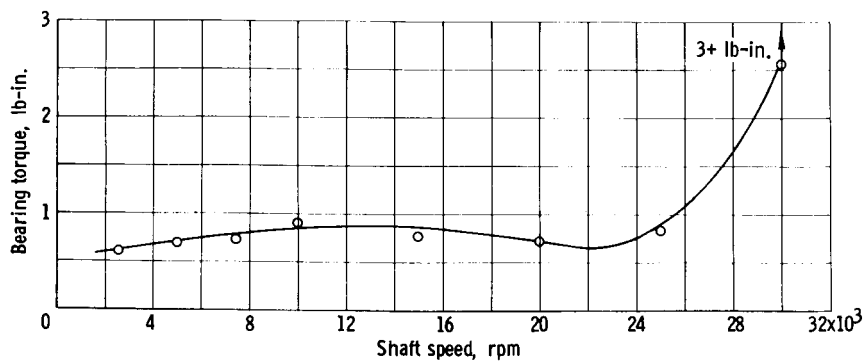stainless steel.

Figure 8. – Experimental bearing torque plotted against shaft speed for 1908-series bearing
operating in liquid oxygen. Bearing, R-11; retainer material, molybdenum disulfide plus
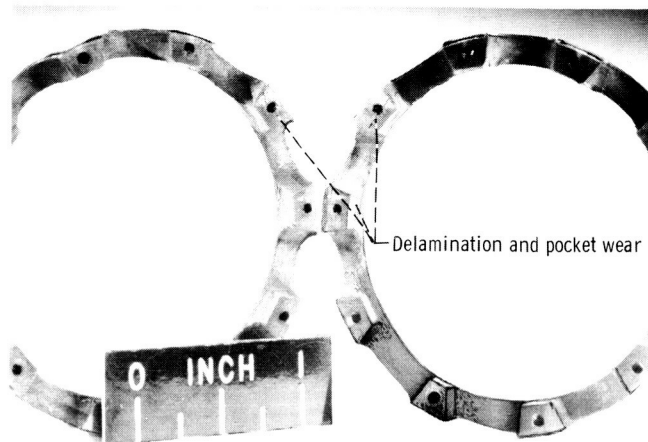glass fiber in Teflon; radial load, 100 pounds.
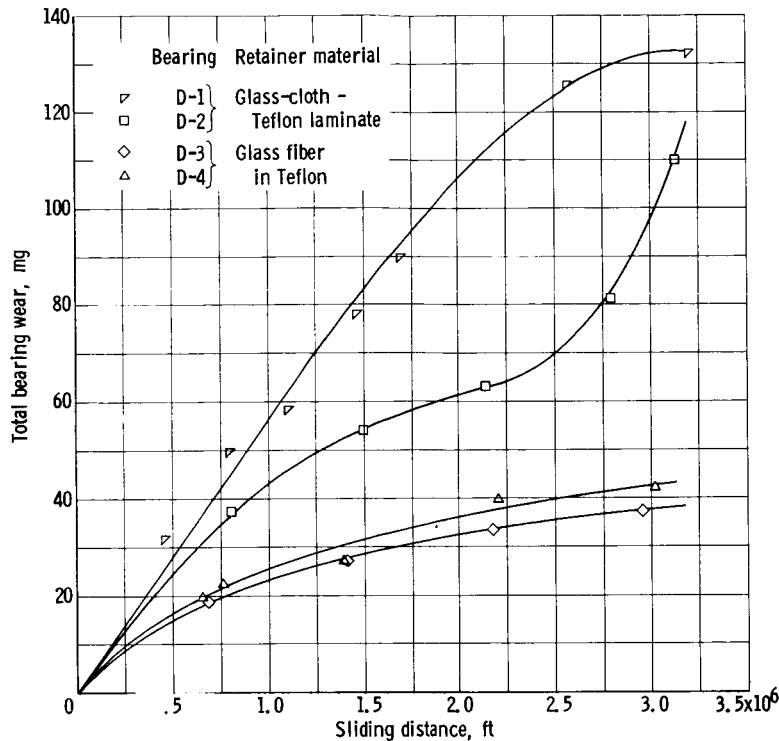
Figure 9. - Total bearing wear plotted against retainer sliding distance for 108-series deep-groove ball bearings at radial loads from 50 to 550 pounds.

30-Percent bronze powder in Teflon. - Performance of this material in bearing R-9 was extremely poor under even the lightest loads. High torques persisted from startup to a speed of 7500 rpm where the test was terminated.

Molybdenum disulfide plus glass fiber in Teflon. - Performance of this material in bearings R-10 and R-11 was generally poor and in direct contrast to results obtained in the 108-series bearings. A load of 200 pounds at the maximum DN value of 1.2 million produced excessive friction torque, and this test was terminated after only 10 minutes. The torque-speed curve of figure 8 shows the steep rate of torque increase at speeds exceeding 25 000 rpm and a 100-pound load. This poor performance is attributed to the method of fabrication and the light construction that allowed the retainer body to move relative to the shroud material under the action of high cage forces. High torques were produced when the ball set came into contact with the aluminum shroud.

100-Percent graphite. - Torque values for bearing R-12, measured during the initial 100-pound load test did not exceed 0.75 pound-inches at speeds to 25 000 rpm. At 30 000 rpm the torque increased to 1.8 pound-inches and remained at this value throughout a 30-minute run. At the completion of this test, it was observed that the retainer had sustained extensive damage at the locating surface and in the ball pockets. Therefore, no further tests were conducted on this material.

Cotton-cloth phenolic laminate. - The rate of torque increase with speed was very steep through 25 000 rpm at a 100-pound load for bearing R-13. At

C-62979

(a) Bearing D-2; retainer material, glass-cloth – Teflon laminate; radial-
load range, 200 to 500 pounds; time at maximum DN (1.2 million), 4.9
hours; total test time, 9.3 hours.
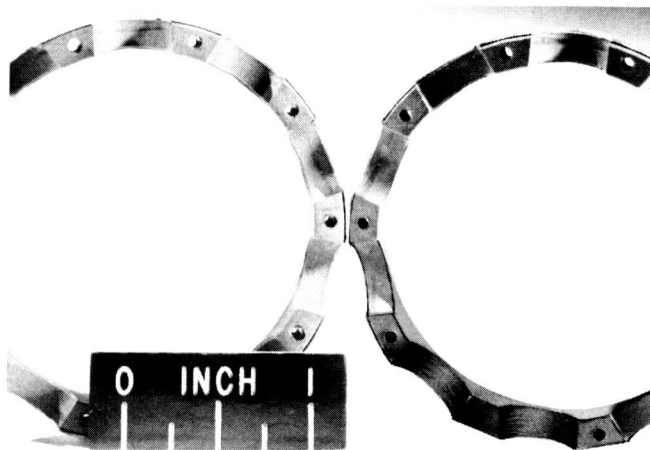
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(b) Bearing D-4; retainer material, glass fiber in Teflon; radial-load range,
100 to 550 pounds; time at maximum DN (1.2 million), 4.5 hours; total
test time, 15.9 hours.

Figure 10. - Extralight 108-series deep-groove ball bearings operated in
liquid oxygen.

27 500 rpm, the bearing seized. Posttest inspection of the oxidant-chamber and
bearing-adapter parts indicated a fire had taken place that completely consumed
the unshrouded retainer. Only the rivets used to hold the two retainer halves
together were found. Races and balls showed evidence of excessively high tem-
peratures. Most of the balls in the set were plastically deformed. This was
the only test where ignition had occurred, and also, the only test where an
oxidizable material was used as a bearing component.

Compacted molybdenum disulfide. - Bearing torque for bearing R-14 did not
exceed 0.75 pound-inches at a 100-pound radial load until a speed of 27 500 rpm
was reached. At this speed the rate of torque increase was rapid, and after
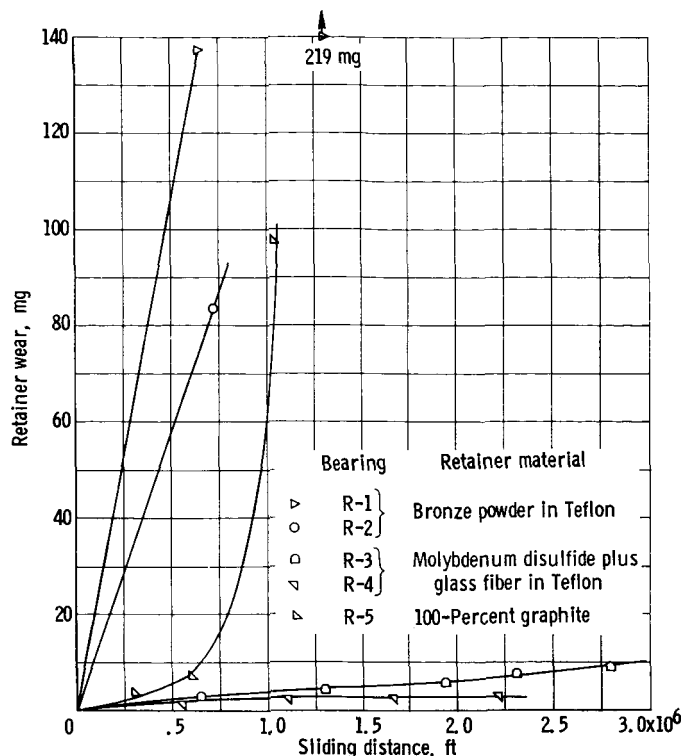25 minutes at 30 000 rpm the torque exceeded 3 pound-inches.

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Figure 11. - Retainer wear plotted against retainer sliding distance for 108-series, radial-separable ball bearings at radial loads from 200 to 600 pounds.
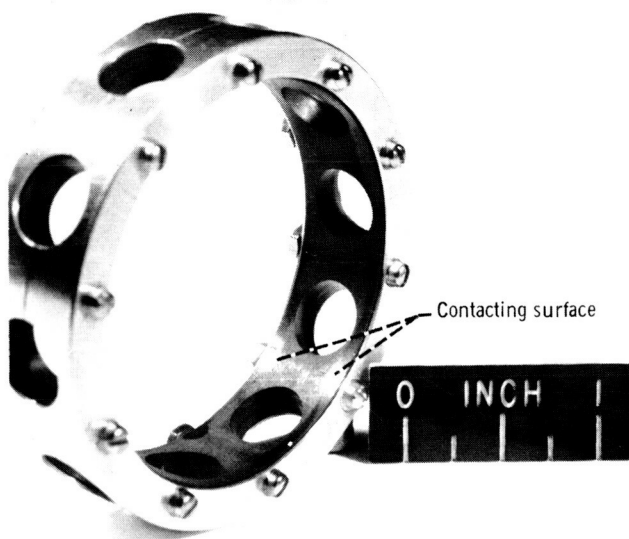
Effect of Total Running Time on Wear of

Extra-Light 108-Series Bearings

Nonseparable or deep-groove type. - Figure 9 is a plot of the total bearing wear against the total sliding distance of the retainer locating surface. Sliding velocities varied from 218 feet per minute at 750 rpm to 8730 feet per minute at 30 000 rpm. It should be pointed out that, for the data of figure 9, the radial load was not constant but varied from 100 to 600 pounds.

The retainers for bearings D-2 and D-4 are shown in figures 10(a) and (b). Bearing D-4 with the glass-fiber-in-Teflon retainer was operated in liquid oxygen over $1\frac{1}{2}$ times as long as bearing D-2 with the glass-cloth - Teflon-laminate retainer, and yet it remained in operable condition. The glass-cloth retainer showed delamination between pockets and extensive wear or deformation of the pockets (fig. 10(a)).

Radial-separable types. - The weight losses plotted in figure 11 against sliding distance are for the retainers only. The retainers made of 30-percent bronze powders in Teflon used in bearings R-1 and R-2 had very high wear rates. At only 5×10^5 feet of sliding the average retainer wear for the two bearings with bronze-powder-in-Teflon retainers (fig. 11) was significantly greater than for the glass-cloth - Teflon retainers (fig. 9). Much of the wear in bearings R-1 and R-2 appeared to have taken place in one area of the locating surface.
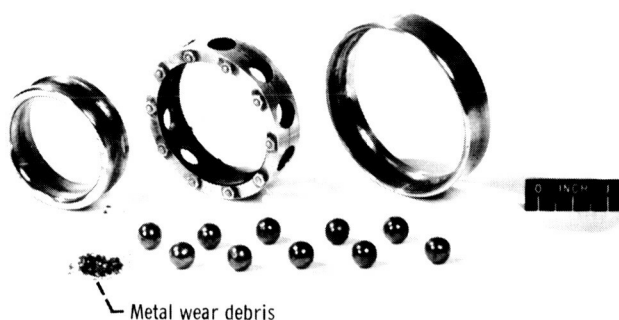
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Contacting surface

O INCH 1

C-63397

(a) Bearing R-3; retainer material, molybdenum disulfide plus glass fiber in Teflon; radial-load range, 200 to 600 pounds; time at maximum DN (1.2 million), 3.6 hours; total test time, 8.5 hours.

Metal wear debris

C-63398

(b) Bearing R-5; retainer material, 100-percent graphite; radial-load range, 200 to 600 pounds; time at maximum DN (1.2 million), 1.2 hours; total test time 3.2 hours.

Figure 12. - Extralight 108-series radial-separable ball bearing operated in liquid oxygen.

This uneven wear is attributed to the nonuniform dispersion of bronze in the Teflon, which resulted in a dynamic unbalance that became significant at high speed.

Bearing R-3 in figure 12(a) illustrates the excellent condition of the retainer after 3.6 hours at the maximum DN of 1.2 million and a total test time of 8.5 hours. The retainer used in bearing R-4 was also in excellant condition at the conclusion of 6.3 hours of testing. The outer races and the ball sets gained weight, which indicated that a transfer of the retainer-body material had taken place. The low wear rate for the molybdenum disulfide plus glass in Teflon used as a retainer in bearings R-3 and R-4 and plotted in figure 11 was predicted by earlier friction and wear studies in liquid nitrogen (ref. 1). No evidence of corrosion or iron oxide deposits was present on the retainer-body material. This contrasts with the other two glass-and-Teflon compositions, which showed evidence of corrosion.

A 100-percent-graphite retainer body (aluminum shrouded) used with test bearing R-5 showed a relatively low wear rate up to 1×10^6 feet of sliding. The test series was concluded at this point because of a failure of the bearing during a test at a 600-pound load. Extreme heating of the balls and races was evident, which probably caused a loss of internal clearance in the bearing and resulted in seizure. Severe damage occurred to both inner and outer races and to the ball set. Considerable metal debris was removed from the test chamber and can be seen in figure 12(b). The retainer experienced material breakout at the ball pockets, and deep scoring caused by
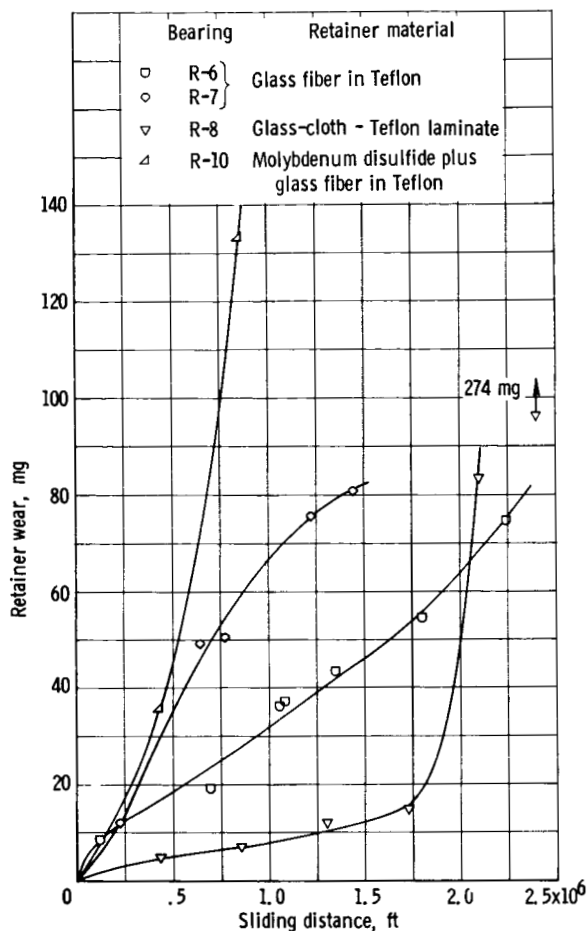
Figure 13. – Retainer wear plotted against retainer sliding distance for 1908-series radial-separable ball bearings at radial loads from 100 to 300 pounds and thrust load of 25 pounds.

metal debris being dragged into the retainer clearance area was evident at the locating surface.

Effect of Total Running Time on

Wear of Extremely Light

1908-Series Bearings

Plots of retainer weight losses against the sliding distance of the locating surface for three different materials can be seen in figure 13. A glass-cloth - Teflon-laminate retainer used with bearing R-8 had the lowest measured wear out to 1.7×10^6 feet: these tests were all at 100-pound loads, and the wear rate was very high. Accompanying this high retainer wear was a correspondingly high weight loss measured from the ball set. At the conclusion of these high-load tests a reddish-brown deposit covered the bearing adapter parts and walls of the test chamber. It is believed that the deposits were extremely fine metallic particles abraded from balls and races by the glass in the 80-percent-glass-cloth retainer body. These fine particles were rapidly oxidized in the surrounding oxygen environment. Samples were generally too small for chemical analysis, and X-ray diffraction patterns obtained were inconclusive. Figure 14(a) shows extensive pocket wear on the retainer of bearing R-8.

The wear rates for the glass-fiber-in-Teflon retainers used in bearings R-6 and R-7 remained fairly constant with no change noted in the R-6 bearing out to 2.25×10^6 feet (fig. 13). Again, as with the glass-cloth - Teflon-laminate retainer, high values of wear were measured on both the balls and the inner race. Varying amounts of the iron oxide were also present on adapter parts, etc. Figure 14(b) shows cracking of the retainer-body material between pockets and between pockets and shroud that occurred in the last 300-pound load test.

The high wear rates measured on the molybdenum disulfide plus glass-fiber-in-Teflon material (bearing R-10, fig. 13 and R-11)) are very much in contrast with the almost negligible wear noted with this material in the extra-light series bearings. It is believed that this inconsistency was not in the material itself but with the difference in retainer construction. The aluminum

Pocket wear
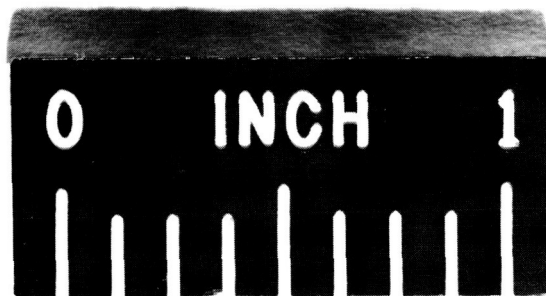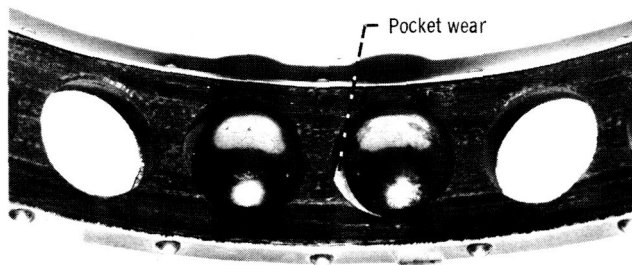
(a) Bearing R-8; retainer material, glass-cloth – Teflon laminate; radial-load range, 100 to 300 pounds; thrust load, 25 pounds; time at maximum DN (1.2 million), 2.4 hours; total test time, 7.8 hours.

Cracks
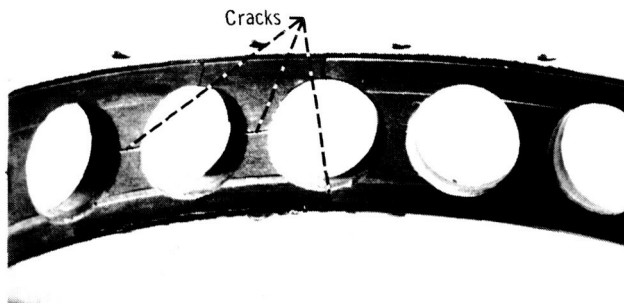
(b) Bearing R-6; retainer material, glass fiber in Teflon; radial-load range; 100 to 300 pounds; thrust load, 25 pounds; time at maximum DN (1.2 million), 3.5 hours; total test time, 11.7 hours.

Damaged shroud

C-73266

(c) Bearing R-10; retainer material, molybdenum disulfide plus glass fiber in Teflon; radial-load range, 100 to 200 pounds; thrust load, 25 pounds; time at maximum DN (1.2 million), 1.2 hours; total test time, 3.9 hours.

Figure 14. – Extremely light 1908-series radial-separable ball bearings operated in liquid oxygen.

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shroud that was spun over the body material was not riveted in place, and movement of the aluminum shroud relative to the retainer body occurred under the action of high cage forces. This permitted the balls to rub against the aluminum instead of the body material. The severely deformed aluminum shroud is evident at each pocket as shown in figure 14(c). Considerable aluminum wear debris, evident at the conclusion of each test, probably made up the largest portion of the total weight loss.

The wear was not recorded for the 100-percent graphite retainer used in bearing R-12 since large pieces had broken out at the locating surface. The results of this test illustrate that the mechanical properties of graphite might limit its use as a ball-bearing retainer material.

The measured weight loss of a compacted molybdenum disulfide retainer after only 8×10^5 feet of sliding was 241 milligrams, over six times the average weight loss of the glass-fiber - Teflon material. The bearing inner-race shoulder indicated a nonuniform contact. The weight loss for the inner race was greater than normally experienced for this short duration.

SUMMARY OF RESULTS

The following results were obtained with extra-light 108- and extremely light 1908-series, 40-millimeter-bore ball bearings running in liquid oxygen to DN values of 1.2 million and radial loads from 100 to 600 pounds:

1. The best overall performance in liquid oxygen for 108-series bearings was obtained with three glass-reinforced-Teflon compositions: these were (1) glass fiber in Teflon, (2) glass-cloth - Teflon laminate, and (3) molybdenum disulfide plus glass fiber in Teflon. The friction and wear results for the glass-reinforced-Teflon compositions corroborate previous work by other investigators in friction and wear studies conducted in liquid nitrogen.

2. The friction torque and wear of extra-light 108-series bearings under these test conditions were superior to those of extremely light 1908-series bearings. These results are in contrast to those reported with liquid-hydrogen-cooled bearings operated at DN values of 1.6 million. The bearings operated in hydrogen, however, were operated under thrust loading, which resulted in less severe retainer forces.

3. Low-torque values and almost negligible wear were observed with molybdenum disulfide plus glass-fiber-in-Teflon retainers used in 108-series ball bearings. All bearing components were in excellent condition after total test times of 8.5 and 6.3 hours on two test bearings.

4. Overall coefficients of friction obtained for bearings equipped with glass-fiber-in-Teflon retainers were of the same order of magnitude as those obtained by other investigators with oil-mist-lubricated bearings of a similar size over the same radial-load range.

5. With 1908-series ball bearings, high rates of torque increase at shaft speeds exceeding 25 000 rpm were observed. The high torques were probably due

to distortion of lightly constructed retainers by high cage forces, characteristic of bearings under combined loading.

6. Torque values did not vary significantly with speed in the 108-series bearings. Generally, the curves are relatively flat as is the case with oil-lubricated rolling-element bearings.

7. High wear rates were observed for bronze in Teflon and compacted molybdenum disulfide retainer materials. Balls and inner races showed relatively high wear in bearings using the glass-fiber-in-Teflon and glass-cloth - Teflon-laminate retainer materials. Retainers of 100-percent graphite evidenced structural failure that would limit its use as a retainer material in cryogenic applications.

Lewis Research Center,
  National Aeronautics and Space Administration,
    Cleveland, Ohio, November 24, 1964.

REFERENCES

1. Wisander, D. W., Maley, C. E., and Johnson, R. L.: Wear and Friction of Filled Polytetrafluoroethylene Compositions in Liquid Nitrogen. ASLE Trans., vol. 2, no. 1, 1959, pp. 58-66.

2. Scibbe, H. W., and Anderson, W. J.: Evaluation of Ball Bearing Performance in Liquid Hydrogen at DN Values to 1.6 Million. ASLE Trans., vol. 5, 1962, pp. 220-232.

3. Butner, M. F., and Rosenberg, J. C.: Lubrication of Bearings with Rocket Propellants. Lubrication Eng., vol. 18, no. 1, Jan. 1962, pp. 17-24.

4. Fogg, A., and Webber, J. S.: The Lubrication of Ball and Roller Bearings at High Speed. Jour. Inst. Petroleum, vol. 39, no. 359, 1953, p. 760.